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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

A Method for Testing
High Temperature
Performance of Greases

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LUBRICATION

A TECHNICAL PUBLICATION DEVOTED TO THE SELECTION AND USE OF LUBRICANTS

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A Method for Testing High Temperature Performance of Greases

MANY years ago when petroleum products included chiefly kerosene and a few heavier lubricating oils, the petroleum chemist discovered that the latter could be thickened by mixing with soaps. This was a happy discovery because it furnished a lubricant which would remain longer in the comparatively large bearing clearances that prevailed, with less chance of dripping out to foul the machinery and create a slippery hazard on the floor. The term grease —long applied to animal fats —was likewise applied to this petroleum compound.

Since those days—scarcely a lifetime ago — machine design and petroleum refining have come a long way. The machinery builder today is continually working to closer tolerances and towards precision finishing of machine parts. The petroleum chemist has adopted a multitude of scientific procedures to guide him in refining and compounding of lubricants which will enable these precision machines to develop the mileage of the modern motor car; to maintain air, rail, marine and highway transport timetables; to produce the necessities of our modern civilization.

So the petroleum scientist evaluates his lubricants under the prospective operating conditions, before he presents these products to the customer. It is necessary to be sure they will perform as expected. The procedure in evaluating the high temperature performance of a modern grease is a case in point. How it is carried out is of interest.

GREASE, the versatile lubricant, has risen to an enviable position in the Petroleum Industry. In the attainment of this status, grease research has paralleled bearing industry research, aircraft and automotive industry developments and instrument builders' designs. In thus keeping up with the trends in industry, grease has contributed prominently to the development of machinery capable of furnishing more output for the power input, with greater dependability, over wider ranges of temperature.

Any grease, of course, must be prepared to function satisfactorily under the expected operating conditions. The laboratory anticipates this by using full scale bearing test devices.

In this issue we present details of a method of test for evaluating high temperature greases. The objective in working out the test procedure was to simulate operating conditions as closely as possible attempting to correlate laboratory testing with actual field practice when using conventional bearings.

Scarcely ten years ago greases had to be adaptable only to a temperature range of between about minus 10°F. and plus 250°F. Military requirements and the aviation industry called for lowering the minimum service temperature requirements to a specified 67°F. below zero and for some special installations to 100°F. below zero. Greases were developed to meet those requirements giving trouble-free operation under the lowest temperature conditions encountered.

At the same time, special requirements of military equipment and the development of such improvements as silicone insulation for electric motors extended the desired upper temperature range for greases to at least 300°F. The military specified a

Figure 1

minimum useful grease life of three hundred hours when used in an anti-friction bearing operating at 10,000 r.p.m. and at a temperature of 300°F. Today the goal is a grease service life of 1000 or more hours under the same conditions. In addition, scientific investigators and designers of gas turbines, turbo-jets, and other machinery are looking forward to the availability of satisfactory lubricants for operation at temperatures as high as 400°F. to 500°F. at high rotative speeds. Such a lubricant must also possess extended service life so that the period between overhauls can be extended.

This widening of the desired operating temperature range assumes added significance when it is realized that greases which might perform satisfactorily for thousands of hours at 250°F. might last only a few hundred hours at 300°F., and further temperature increases reduce the service life even more drastically as will be shown later in this article.

There used to be a time when appearance, feel, and penetration were the criteria by which greases were judged. That is no longer true. Today, each new grease developed, whether it be an improved product for ball and roller bearings or "tailored" for some other specific application, must be evaluated under conditions that simulate those under which it will be required to function in service. The petroleum industry has dovetailed grease development with chemical and mechanical evaluations not only to determine specific grease properties but also to guide the research chemist in his development work on new products. This teamwork has been exemplified during the development of greases for high temperature-high speed applications.

The dependability with which modern greases designed for this service are functioning is evidence of the value of the laboratory work which has attended their preparation. With the knowledge that such greases are available, the reader should be interested in knowing how the laboratory makes sure that they are suited to his particular operating conditions.

HOW THE PROBLEM WAS APPROACHED

As is well known to all who have had experience with research, the method of approach in solving any new problem must of necessity go through a transitory status. From an economic point of view, it is always advisable to look into the applicability of existing machinery before developing new apparatus. Very often it is possible to arrange a combination of operating mechanisms and measuring instruments to good advantage. This becomes espepecially practicable in any study involving ball or roller bearings where the degree of lubrication from

any type of lubricant is to be studied under extremes of speed and temperature.

This procedure was followed in the preliminary investigational work on the subject under discussion, i.e. high temperature performance testing of greases. Certain fundamentals pertaining to design and installation soon became evident from early work.

1 — Each individual test unit should be mounted in a firm installation to avoid excessive vibration.

2 — The main shaft carrying the test bearing and driving motor connections should be as short as possible to eliminate "shaft whip."

3 — This shaft should be of hardened steel to reduce the wear induced through assembly and disassembly of the unit.

4 — Adequate provision should be made for even heat distribution around the test bearing. This is particularly important where bearings and lubricants are to be tested under high temperature conditions.

All the foregoing pointed to the necessity of designing a special machine. This unit was planned so as to eliminate as far as possible any mechanical factors that might contribute to machine failure and thereby possibly disqualify an experimental grease that looked good, or so confuse the results on a planned series of experimental products that no valid conclusions could be drawn. Ideally, a mathematical representation of the test would indicate the machine factors as constants with only one variable present, the grease under test.

DETAILS OF THE MACHINE

Figure 1 shows three test units set up with connections to the panel board. A test unit consists essentially of a test bearing mounted on a spindle which in turn is supported on suitable anti-friction bearings in a housing and driven by an electric motor by means of a flat belt. Figure 2 is a cross-section view which shows the test unit held in the pedestal and surrounded by the heater chamber. The pedestal is mounted on top of a firm base with the drive motor mounted on a pivoted base below. A flat woven linen belt rides on the motor and spindle drive pulleys.

Type of Spindle

The spindle on which the test bearing is mounted is made of a high grade alloy steel and heat treated after rough grinding to relieve all stresses. The diameters are held to within one ten-thousandth of an inch concentricity and the spindle is dynamically balanced so as to preclude vibration. The distance between the support bearings is small and the overall length is under eleven inches. These distances are held at a minimum to further reduce vibration and shaft whip at the desired speed of 10,000 r.p.m.

Fretting Corrosion Eliminated

Previous experiences with high speed-high temperature test machines indicated considerable trouble with fretting corrosion of the spindle surfaces which the test bearings contacted. In order to obviate this factor, the surfaces on which the test bearings are pressed have been chrome-plated. This was done by grinding the surfaces 0.002 inch undersize, chrome-plating to 0.003 inch oversize, and then grinding to size. This procedure has effectively eliminated fretting corrosion of the surfaces over a period of two years.

Another feature of the spindle design is the

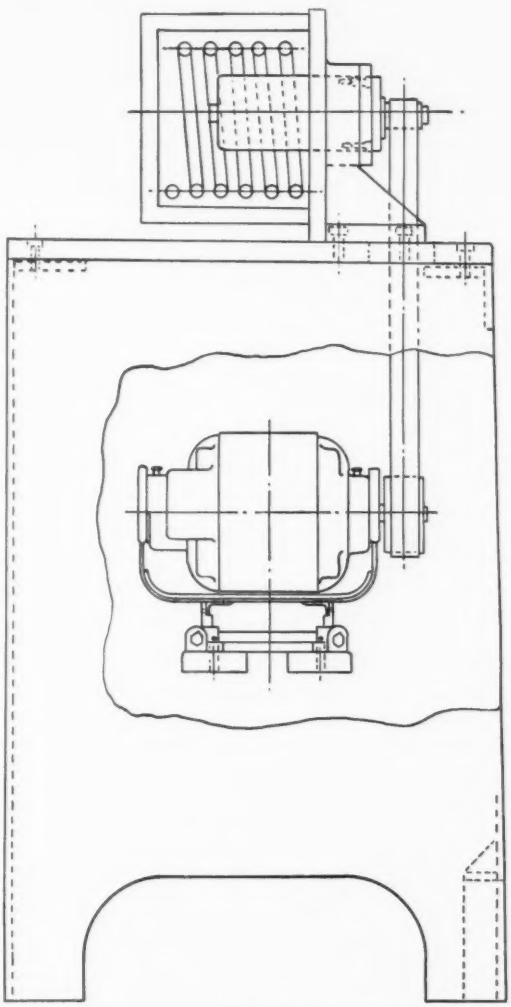

Figure 2

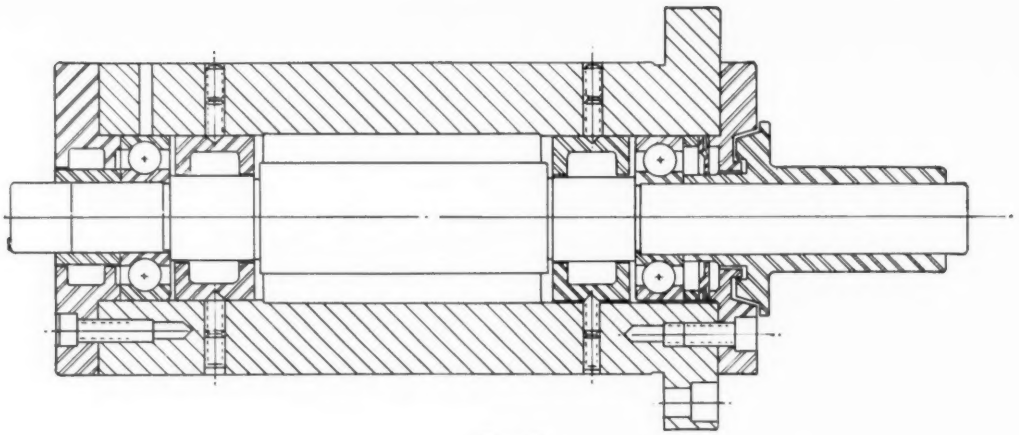

Figure 3

use of press-fits for the bearings, pulley, etc. to eliminate unbalance which might occur if locking devices were used. Here again the chrome-plated surfaces have proven exceptionally resistant to wear over a long period of time during which the units were assembled and disassembled quite frequently.

Figure 3 is a cross-section of the spindle and housing assembly while Figure 4 is an exploded view showing the individual parts. The bearings, grease slingers or shields, pulley, and spacer rings are assembled on the spindle before inserting it into the housing.

Housing Details

The spindle housing has an overall length of eight inches and a diameter of 3.5 inches. The housing has been provided with a 0.75 inch wall, for two reasons; first for rigidity and second for bulk to maintain a more uniform temperature distribution. The housing and cover plates act both as shields and as positioners for the outer races of the bearings. An adjusting spring is interposed between the end cover plate on the pulley end and the spacer ring which butts against the outer race of the outboard bearing.

Loading

The clearances are such that a thrust load of 17.5 pounds is applied to the bearings. This load is applied only to maintain contact between the balls and races of the bearings when operated at 10,000 r.p.m.

No radial load is imposed on the test bearing other than that of the drive motor and belt tension caused by the angle of wrap on the small pulley. The total load is relatively small and is maintained as nearly constant as possible. The primary purpose in operating the test under no load, other than that sufficient to keep the balls and races in constant contact, is to hold the load as far as possible below the rated capacity of the bearings. The input of heat to the bearing must necessarily set up stresses in the bearing. Such stresses, if coupled with a load even approaching the capacity of the bearing, would bring the bearing back into the picture as a factor contributing to failure and reduce the sensitivity of the test as a grease evaluation mechanism.

Function of the Shields

Although the housing cover plates serve as outside shields for the bearings, individual shields are provided for the inside. The inside shields have an annular space which functions as a trap to prevent oil or grease from migrating from the bearings to the shaft. These shields are carefully positioned and held in place by set screws in the housing. A slight clearance between the test bearing cover plate and the annular ring which holds the inner race of the bearing tightly against the shoulder of the spindle provides means for the escape of volatilized oil or oxidation products from the grease. It also enables a certain amount of "breathing" by the bearing, a condition which is usually but not always present in service.

Temperature Measurement

The measurement and recording of bearing temperatures is an important factor in this type of test since, as mentioned previously, slight variations in temperatures in this range have a pronounced effect on the grease life. A hole in the housing wall, shown in Figure 3, provides for the installation of a special type iron-constantan thermocouple. This thermocouple is a pencil type with its thermal junction in the tip. When in position, the thermal junction of the thermocouple contacts the periphery of the outer race of the test bearing. The temperatures measured are automatically recorded by an instrument mounted on the panel board.

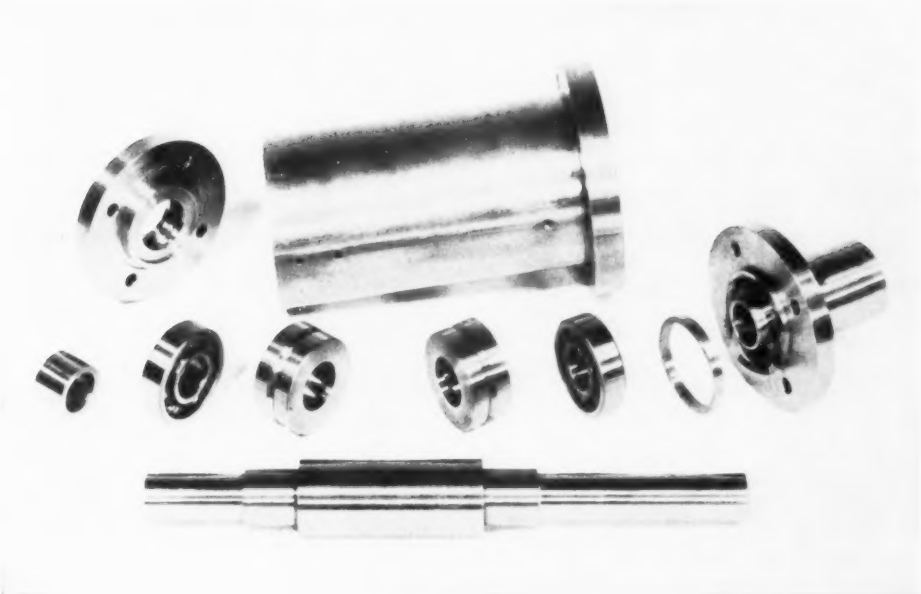

Figure 4

Both the test and outboard bearings are ABEC* No. 3 Grade 204 ball bearings specially processed for high speed high temperature operation and are stress-relieved at from 350°F. to 400°F.

The test bearing housing bolts into a semi-circular pedestal, 2.5 inches wide and designed so that when the housing is in position its centerline stands six inches above the top of the base. This pedestal also has a wide rim to which the bearing housing as well as the heater chamber are bolted.

Two heaters in the heater chamber provide sufficient heat to run the test bearing at temperatures up to and including 500°F. One heater has a rating of 1000 watts and the second 500 watts; both are 230 volt Calrod tubular heaters bent to conform to the heater chamber. The 1000 watt heater provides a constant heat and is controlled through a variac. Under test conditions the constant heater provides from 80 to 90 per cent of the required heat. The 500 watt heater operates intermittently to maintain the balance of the heat required. A DeKhotinsky thermo-regulator in the heater chamber, adjacent to the bearing housing, regulates the heat input of the intermittent heater. Temperature regulation has held within relatively close limits (± 5°F.) and test bearing temperatures up to and including 475°F. have been maintained. The assembly and disassembly of the test unit can be accomplished without disturbing the heaters or DeKhotinsky heat controller.

The Spindle Drive

A 0.5 horsepower, 3450 r.p.m. 230 volt cradle-mounted electric motor drives the test spindle. This motor, as mentioned previously, rests on a pivoted base below the test unit and is belted to the test spindle pulley by a flat woven belt 1.25 inches wide. In order to reduce vibration to a minimum, the drive pulley on the motor was dynamically balanced and the rotor of the motor also checked for balance.

The ratio of diameters of drive pulley on the motor and pulley on the test spindle is approximately 2.9 to 1 for a speed of 10,000 r.p.m. However, the spindle speed was checked with a strobotac and adjustments made to bring the speed within that required. Belt slippage proved to be very low. Incidentally, both pulleys have a 2° crown. Various controls were tried in the motor circuit. However, the most satisfactory arrangement proved to be a heavy fuse in the starting circuit and a fuse somewhat below the full load requirement in the running circuit coupled through a double pole, double throw switch with neutral postion.

The motor is started with the high amperage fuse circuit and, when speed is attained, the switch is thrown to the low amperage or running circuit. The rupture of the low amperage fuse when the load on the motor exceeds the fuse capacity determines the failure point of the test. Noises developed in the test bearing and temperature surges are observed and recorded, but are not considered in determining the failure point, although they may be indicative of imminent failure.

TEST PROCEDURE

The test procedure evolved for evaluating greases at high temperatures consists in charging 3.0

*Annular Bearing Engineers Com.

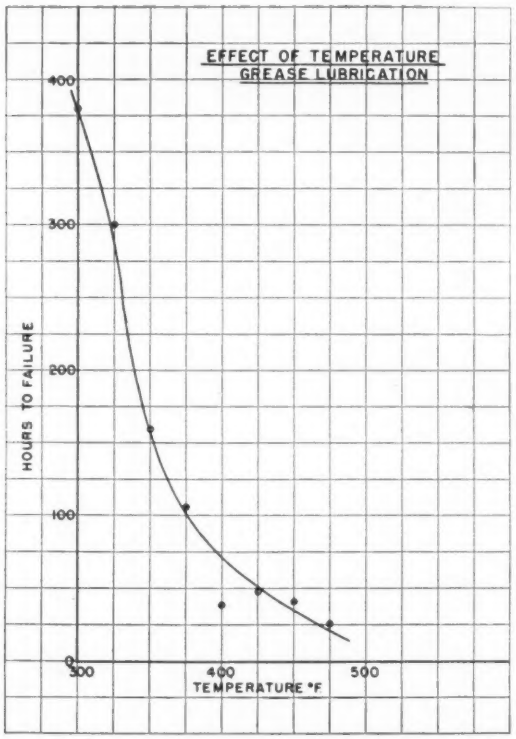

Figure 5

(±0.1) grams (approximately 50 per cent of full charge) to the test bearing with a narrow-blade spatula and working the grease well into the interstices of the bearing. The charged bearing is then rotated clockwise and again counterclockwise at 200 r.p.m. for one minute each to work the grease into the bearing. The outboard bearing, which does not operate in the high temperature zone, is lubricated with a high quality high temperature grease. The unit is now ready for assembly.

The unit is operated in cycles as follows: The motor is started and the heaters turned on simultaneously to bring the bearing temperature up to its predetermined level where it is maintained for the duration of the cycle. The machine is run for twenty-four hours counted from the start. A shutdown and cooling period of two hours follows the high temperature portion. This cycle is repeated until failure occurs.

The actual effect of the cooling period has not been fully determined. Judged by the number of failures on starting, it appears that a shellac-like formation on the races of the bearing hardens enough during the cooling period to cause skidding of the balls and seizure. The shellac-like formation, although present at the high temperatures, is apparently in a more plastic state at those temperatures and does not seriously hamper operation.

Reproducibility of Test

A series of twenty-nine tests on eight different units was made using an uninhibited experimental grease of known performance characteristics on other high temperature test machines. This particular product did not have a very high useful life and by its use the time required for reproducibility studies was greatly shortened.

The data on these twenty-nine tests are presented in Table I:

Unit No.	*Hours to Failure*	*Bearing Condition After Test*
1	182	Slightly rough
1	240	Seized
1	216	Seized
1	216	Very rough
2	206	Seized
2	120	Very rough
2	238	Very rough
2	216	Seized
3	230	Seized
3	214	Seized
3	240	Very rough
4	168	Seized
4	384	Seized
4	241	Seized
4	243	Very rough
5	143	Seized
5	168	Seized
5	144	Very rough
5	336	Seized
5	264	Seized
6	216	Seized
6	240	Seized
7	120	Seized
7	408	Seized
7	192	Seized
7	237	Seized
8	194	Seized
8	145	Seized
8	216	Seized

The average time to failure for the twenty-nine tests was 220 hours.

Twenty-one of the twenty-nine tests included in the average were within plus or minus two cycles of operation while 25 were within three cycles. The remaining tests were four cycles above or below the average.

It is realized that this is not a true average as a minimum of nine or ten tests on each unit would be necessary in order to more nearly approach it. The results obtained, however, are considered to be within the limits of good reproducibility for this type evaluation. The determination of a true average would not be warranted due to the time and materials required.

Effect of Temperature

The next logical step in the study of greases under high temperature conditions was to ascertain the effect of elevated temperatures, that is, above 300°F., on their lubrication characteristics. A grease that had a useful life of 380 hours at 300°F. and 10,000 r.p.m. on the High Temperature Performance Test was selected for the high temperature investigations. Figure 5 graphically represents the data obtained from a series of tests where the bearing temperature was raised from 300°F. to 475°F. in increments of twenty-five degrees. The useful life of the grease as indicated by hours to failure time shows a very rapid decline with an increase of 50°F. from 300°F. to 350°F. At 375°F. the useful life has been further reduced to approximately 25 per cent of the original life. Each further increase in temperature decreases the useful life until at a temperature of 475°F. the maximum time is one cycle of operation.

A large number of products have been evaluated at 450°F. This number included greases made with different soaps and blended with various viscosity oils, both mineral and synthetic.

The reproducibility of mechanical tests on greases is rather difficult and especially so at high rotative speed and high temperatures. Perhaps one of the greatest contributing factors is the disposition of the grease on the bearing when it has attained maximum speed. It is highly improbable that the same amount of grease will be retained on the bearing races and ball cage, or that the same amount of grease will be thrown out of the bearing to the shields in each of a series of tests on the same lubricant.

The bearings used for test purposes appear to be quite satisfactory. However, it is impossible to state definitely at this time just how much variation in the bearings contributes to variation in failure time. This is especially true in the 300°F. to 450°F. bearing temperature range. Furthermore, the quality of individual bearings for test purposes cannot be determined with any degree of accuracy, although a crude screening test does separate those bearings that are entirely unfit. A program is now in progress to discover a non-destructive method for evaluating bearings more precisely before they are placed on the grease test unit.

INDICATION OF STARVED LUBRICATION

An anti-friction bearing when operating under conditions approaching starved lubrication will develop a high pitched squeal. In the early work on testing greases for high temperature operation this squeal was taken as the failure point. However, examinations of the bearings after test frequently showed them to be in excellent condition and the grease still appeared to be suitable for continued operation. Further investigations revealed that although a noise developed in the test bearing, it would frequently fade out and the bearing would subsequently operate quietly. It was then assumed that when a noise developed, local heating also developed. This in turn drew more of the lubricant into the trouble area and starved lubrication conditions were relieved. Failure, however, is imminent when the noise persists or recurs frequently. A continued noise indicates that the grease itself has lost

Figure 6

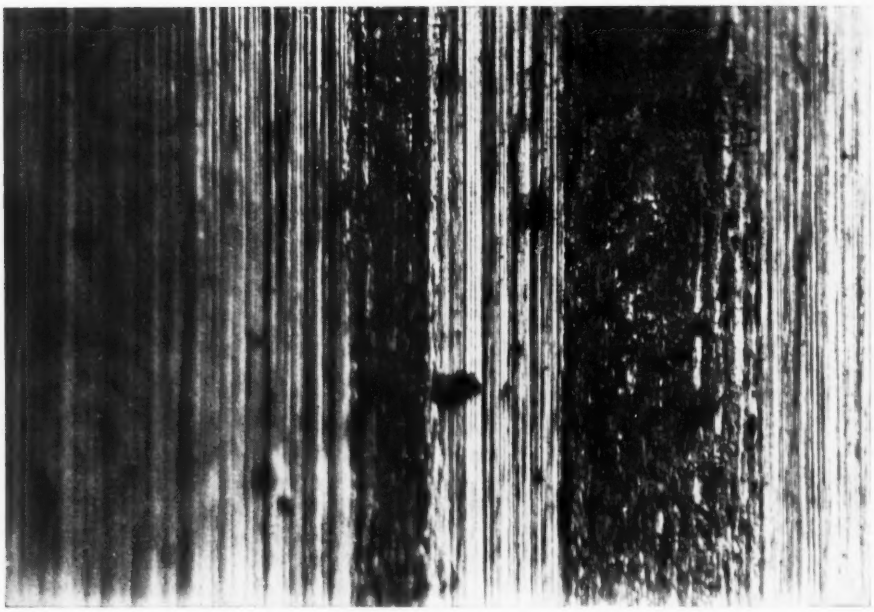

Figure 7

its feeding characteristics and is no longer able to maintain the proper film. The subject of failures will be more fully explained below.

EVALUATION OF PHOTO-MICROGRAPHS

A number of photomicrographs are presented to illustrate the conditions of the grease at failure. All photomicrographs were taken with vertical illumination at 100X magnification and picture the surface of the bearing inner race on both sides of the path traveled by the balls during rotation.

Photomicrograph (Fig. 6) shows the surface of a new unused bearing that was rejected by the screening test. It is evident from the condition of the surface that a surface tension had developed in the metal great enough to cause rupture. This bearing would probably function very satisfactorily under service conditions but not for test purposes.

Photomicrographs shown by Figs. 7 and 8 illus-

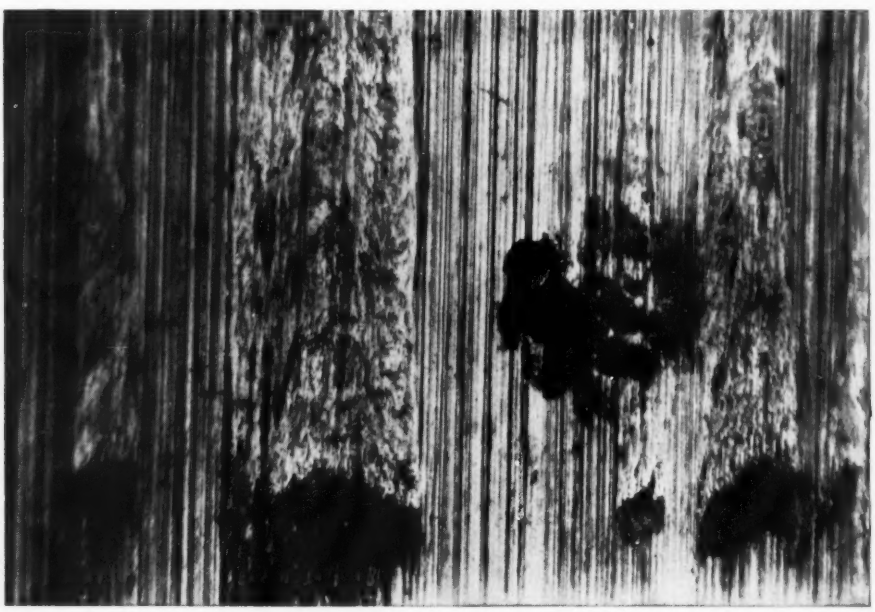

Figure 8

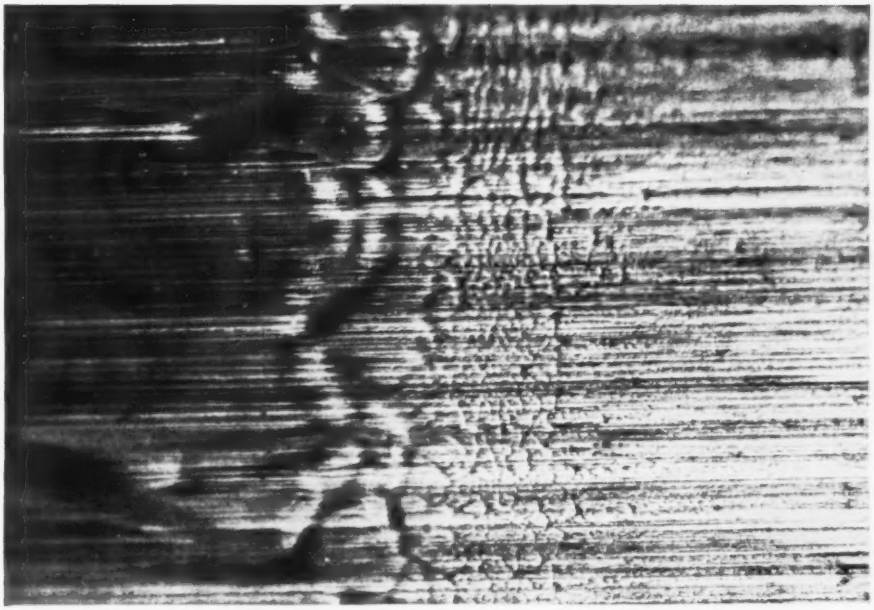

Figure 9

trate two different types of failure. Those failures were developed at ordinary temperatures but under starved lubrication conditions and high rotative speeds.

Two distinct ball paths are seen on Fig. 7. The narrow path developed under conditions of no load whereas the broad path was produced by a light thrust load which moved the ball path off center. The metal surface in the path has become rough but is not gouged.

The type of failure shown by Fig. 8 was also made under no load-starved lubrication conditions and high rotative speed; however, the failure had progressed to a point where the surface shows some flow of metal.

The photomicrograph in Fig. 9 illustrates a condition of good anti-friction bearing lubrication. This was produced by charging a new bearing with grease, assembling the bearing in the High Temperature Performance Test Machine, and rotating it at 10,000 r.p.m. for ten minutes. The bearing at the end of the run was removed and carefully dis-

Figure 10

Figure 11

assembled so as not to disturb the grease. The picture shows a wedge of built up grease followed by a gradually tapering film. The channel produced by the passage of the balls has filled in so as to present an unbroken film.

Figs. 10 and 11 show the surfaces of a bearing inner race that had been in operation for over 500 hours at 250°F. bearing temperature and 10,000 r.p.m. Fig. 11 shows a shellac-like material built up on the surface, while Fig. 10 shows the same surface that had been wiped with a cloth wet with a solvent. Although all of the deposit had not been wiped off, enough was removed to show the metal surface of the bearing. This bearing had seized at failure; consequently, one would expect to find a spalled or broken surface. This type of failure, however, seldom damages the surface of the bearing because it is due to a building up of shellac-like deposit that eventually reduces the clearances in the bearing sufficiently to cause seizure.

Fig. 12 illustrates another type of lubricant failure. Here the surface is covered with an excellent

Figure 12

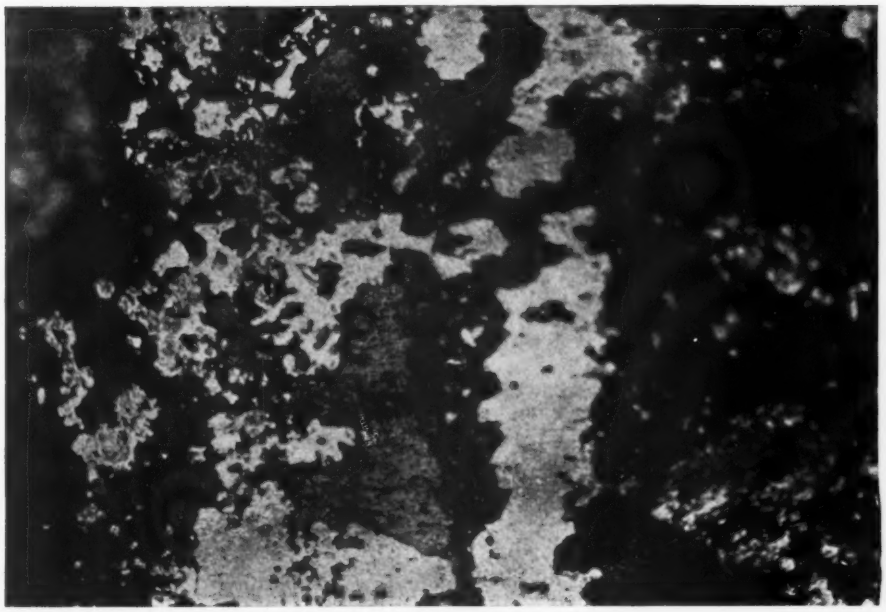

Figure 13

supply of lubricant, yet the bearing which this grease lubricated seized after six hours running at 450°F. and 10,000 r.p.m. An examination of the grease showed it to have developed an extremely tacky texture, which in all probability increased under the conditions of test to such an extent as to prevent the operation of the bearing.

Figs. 13, 14 and 15 illustrate the various stages of the shellac-like deposit on the bearing surfaces as it changes from a shellac to a hard carbon with a gradual covering of the entire surface.

Fig. 16 clearly shows the ball path running across the picture. This ball path has the same type of surface found on the bearing operated under starved-lubrication conditions. The light strip running diagonally across the path was made by a knife blade in chipping off the carbonized material. This particular grease lubricated satisfactorily for 48 hours at 425°F.

All bearings were found to have broken ball retainer cages where seizure occurred at 450°F. and 475°F.

Figure 14

Figure 15

CONCLUSION

Until this latest machinery for high temperature testing of ball and roller bearing lubricants was perfected, the reliability of results obtained from previous devices was questionable. Obviously, reproducibility of results is a most important factor. Unless this is possible, correlation of laboratory procedure with operating service cannot be accomplished. This newest machine has an additional advantage in that, while presently built to operate at 10,000 r.p.m., with but minor changes in construction, it can be speeded up to around 30,000 r.p.m. This availability of a machine for testing anti-friction bearings under high temperature and very high speeds opens another field for further research.

Figure 16

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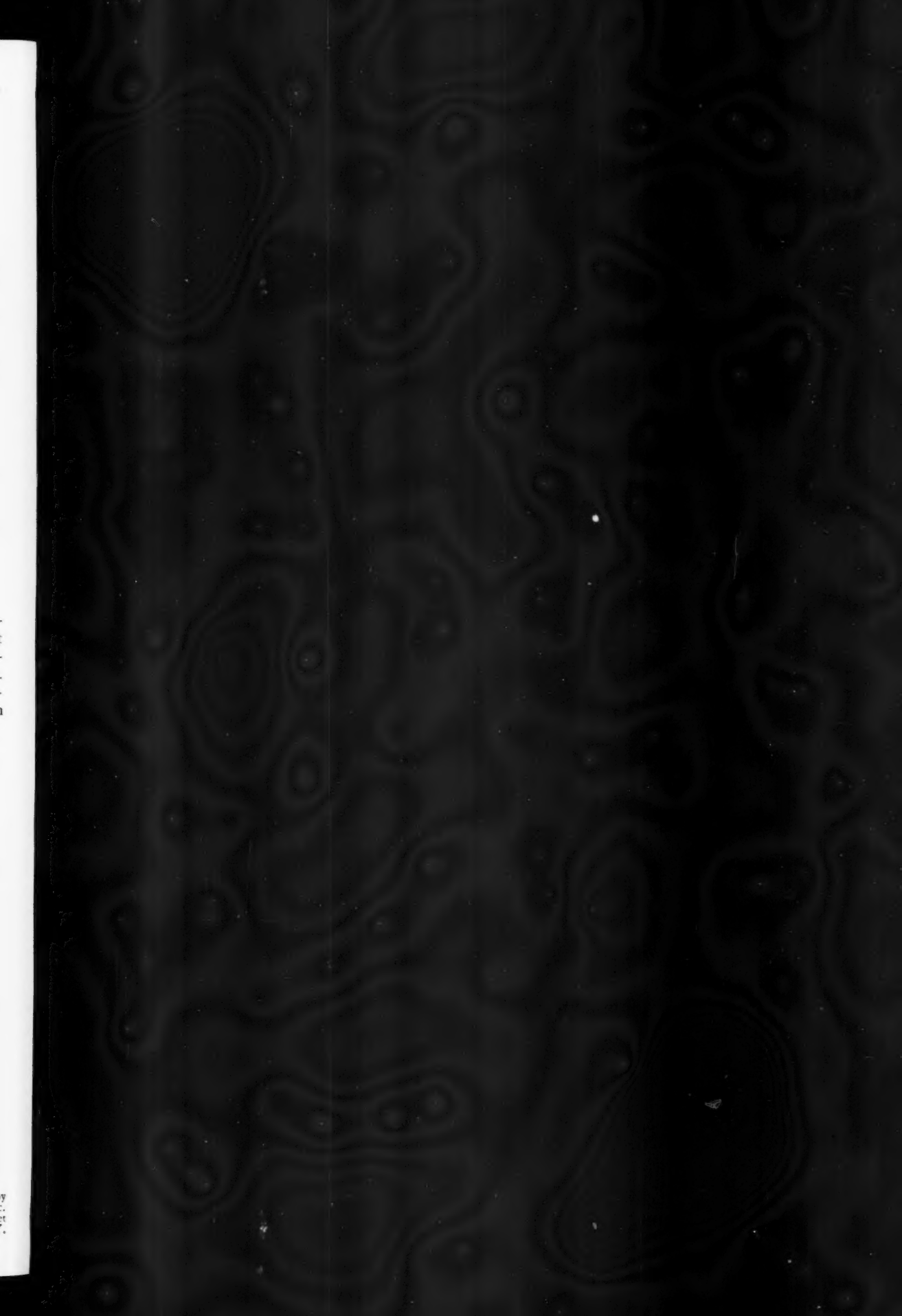

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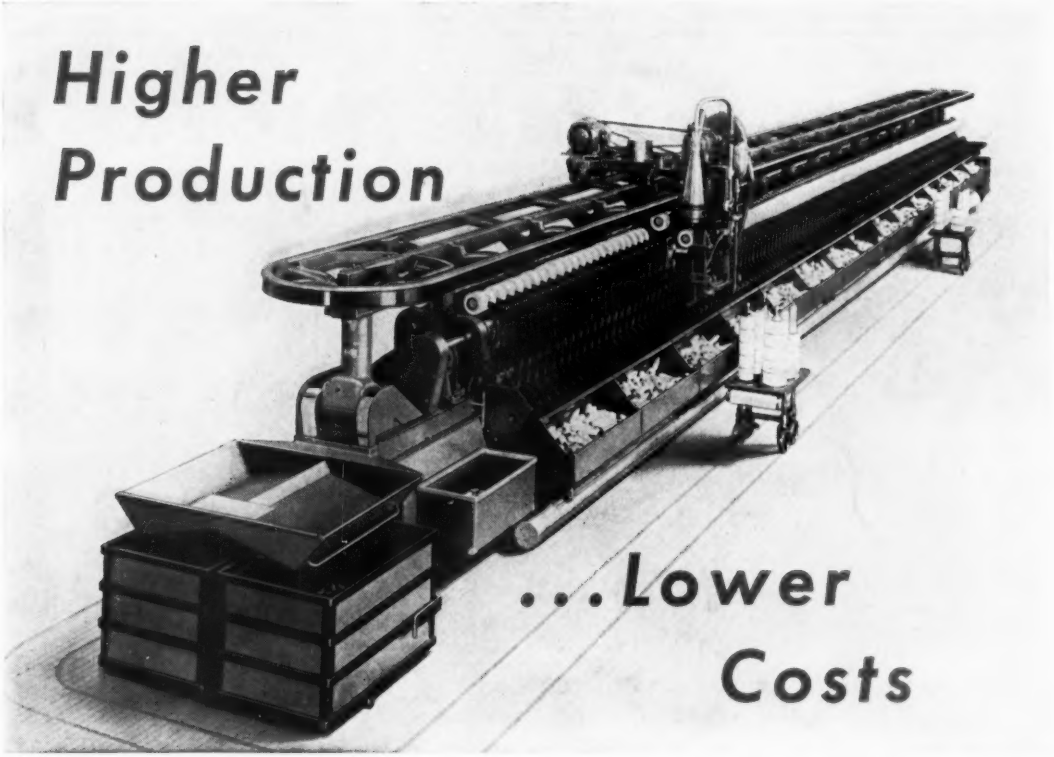

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